OFFSET THREAD SCREW ROTOR DEVICE

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation of U.S. Application No. 10/283,421, filed on October 29, 2002 which is a continuation-in-part of U.S. Application No. 10/013,747, filed on October 19, 2001 and issued as U.S. Patent No. 6,599,112 on July 29, 2003.

This application is also related to the subject matter in co-pending U.S. Application No. 10/283,422, filed on October 29, 2002.

STATEMENT REGARDING FEDERALLY SPONSORED RESEARCH OR DEVELOPMENT

Not Applicable.

BACKGROUND OF THE INVENTION

1. FIELD OF THE INVENTION

This invention relates generally to rotor devices and, more particularly to screw rotors.

2. DESCRIPTION OF RELATED ART

Screw rotors are generally known to be used in compressors, expanders, and pumps. For each of these applications, a pair of screw rotors have helical threads and grooves that intermesh with each other in a housing. For an expander, a pressurized gaseous working fluid enters the rotors, expands into the volume as work is taken out from at least one of the rotors, and is discharged at a lower pressure. For a compressor, work is put into at least one of the rotors to compress the gaseous working fluid. Similarly, for a pump, work is put into at least one of the rotors to pump the liquid. The working fluid, either gas or liquid, enters through an inlet in the housing, is positively displaced within the housing as the rotors counter-rotate, and exits through an outlet in the housing.

The rotor profiles define sealing surfaces between the rotors themselves between the rotors and the housing, thereby sealing a volume for the working fluid in the housing. The profiles are traditionally designed to reduce leakage between the sealing surfaces, and special attention is given to the interface between the rotors where the threads and grooves of one rotor respectively intermesh with the grooves and threads of the other rotor. The meshing interface between rotors must be designed such that the threads do not lock-up in the grooves, and this has typically resulted in profile designs similar to gears, having radially widening grooves and tightly spaced involute threads around the circumference of the rotors.

However, an involute for a gear tooth is primarily designed for strength and to prevent lock-up as teeth mesh with each other and are not necessarily optimum for the circumferential sealing of rotors within a housing. As discussed above, threads must provide seals between the rotors and the walls of the housing and between the rotors themselves, and there is a transition from sealing around the circumference of the housing to sealing between the rotors. In this transition, a gap is formed between the meshing threads and the housing, causing leaks of the working fluid through the gap in the sealing surfaces and resulting in less efficiency in the rotor system. A number of arcuate profile designs improve the seal between rotors and may reduce the gap in this transition region but these profiles still retain the characteristic gear profile with tightly spaced teeth around the circumference, resulting in a number of gaps in the transition region that are respectively produced by each of the threads. Some pumps minimize the number of threads and grooves and may only have a single acme thread for each of the rotors, but these threads have a wide profile around the circumferences of the rotors and generally result in larger gaps in the transition region.

BRIEF SUMMARY OF THE INVENTION

It is in view of the above problems that the present invention was developed. The invention features a screw rotor device with phase-offset helical threads on a male rotor that mesh

with the identical number of corresponding phase-offset helical grooves on a female rotor. Another feature of the invention is the cut-back concave profile of the helical groove and the corresponding shape of the cut-in convex profile that meshes with the cut-back concave profile of the helical groove. The cut-back concave profile corresponds with a helical groove having a radially narrowing axial width at the periphery of the female rotor. Yet another feature of the invention is the buttress thread profile of the helical threads and the helical grooves. Additionally, another aspect of the invention is limiting the maximum length of the rotors to a single pitch of the helical thread and groove. The features of the invention result in an advantage of improved efficiency of the screw rotor device.

Further features and advantages of the present invention, as well as the structure and operation of various embodiments of the present invention, are described in detail below with reference to the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

The accompanying drawings, which are incorporated in and form a part of the specification, illustrate the embodiments of the present invention and together with the description, serve to explain the principles of the invention. In the drawings:

Figure 1 illustrates an axial cross-sectional view of a screw rotor device according to the present invention;

Figure 2A illustrates a detailed cross-sectional view of one embodiment of the screw rotor device taken along the line **2-2** of Figure 1;

Figure 2B illustrates a detailed cross-sectional view of another embodiment of the screw rotor device taken along the line **2-2** of Figure 1;

Figure 3 illustrates a detailed cross-sectional view of the screw rotor device taken along line **3-3** of Figure 1;

Figure 4 illustrates a cross-sectional view of the screw rotor device taken along line **4-4** of Figure 1; and

Figure 5 illustrates a schematic diagram of an alternative embodiment of the invention.

Figure 6A illustrates a detailed cross-sectional view of the screw rotor device taken along line **6-6** of Figure 2A.

Figure 6B illustrates a detailed cross-sectional view of the screw rotor device taken along line **6-6** of Figure 2B.

Figure 7A illustrates an axial cross-sectional view of another alternative embodiment of the screw rotor device according to the present invention

Figure 7B illustrates a lengthwise cross-sectional view of the screw rotor device taken along line **7B-7B** of Figure 7A.

DETAILED DESCRIPTION OF THE INVENTION

Referring to the accompanying drawings in which like reference numbers indicate like elements, Figure 1 illustrates an axial cross-sectional schematic view of a screw rotor device **10.** The screw rotor device **10** generally includes a housing **12**, a male rotor **14**, and a female rotor **16.** The housing **12** has an inlet port **18** and an outlet port **20.** The inlet port **18** is preferably located at the gearing end **22** of the housing **12,** and the outlet port **20** is located at the opposite end **24** of the housing **12.** The male rotor **14** and female rotor **16** respectively rotate about a pair of substantially parallel axes **26, 28** within a pair of cylindrical bores **30, 32** extending between ends **22, 24.**

In the preferred embodiment, the male rotor **14** has at least one pair of helical threads **34,** 36, and the female rotor **16** has a corresponding pair of helical grooves **38, 40.** The female rotor **16** counter-rotates with respect to the male rotor **14** and each of the helical grooves **38**, **40** respectively intermeshes in phase with each of the helical threads **34, 36.** In this manner, the working fluid flows through the inlet port **18** and into the screw rotor device **10** in the spaces **39, 41** bounded by each of the helical threads **34, 36**, the female rotor **16,** and the cylindrical bore **30**

around the male rotor **14**. It will be appreciated that the helical grooves **38, 40** also define spaces bounding the working fluid. The spaces **39, 41** are closed off from the inlet port **18** as the helical threads **34, 36** and helical grooves **38, 40** intermesh at the inlet port **18**. As the female rotor **16** and the male rotor **14** continue to counter-rotate, the working fluid is positively displaced toward the outlet port **20**.

The pair of helical threads **34, 36** have a phase-offset aspect that is particularly described in reference to Figures 2A, 2B and 3 which show the cross-sectional profile of the screw rotor device through line **2-2**, the two-dimensional profile being represented in the plane perpendicular to the axes of rotation **26, 28**. The phase-offset aspect is also discussed below in reference to Figure 7A. The cross-section of the pair of helical threads **34, 36** includes a pair of corresponding teeth **42, 44** bounding a toothless sector **46**. The phase-offset of the helical threads **34, 36** is defined by the arc angle β subtending the toothless sector **46** which depends on the arc angle α of either one of the teeth **42, 44**. In particular, for phase-offset helical threads, the toothless sector **46** must have an arc angle β that is at least twice the arc angle α subtending either one of the teeth **42, 44**. The phase-offset relationship between arc angle β and arc angle α is particularly defined by equation (1) below:

$$\text{Arc Angle } \beta \quad \geq \quad 2 * \text{Arc Angle } \alpha \qquad (1)$$

As illustrated in Figures 2A and 2B, the angle between ray segment **oa** and ray segment **ob**, subtending tooth **42**, is arc angle α. According to the phase-offset definition provided above, arc angle β of the toothless sector **46** must extend from ray segment **ob** to at least to ray segment **oa'**, which would correspond to twice the arc of arc angle α, the minimum phase-offset multiplier being two (2) in equation 1. In the preferred embodiment, the arc angle β of the toothless sector **46** extends approximately five times arc angle α to ray segment **oa"**, corresponding to a phase-offset multiplier of five (5). Accordingly, another two additional teeth could be potentially fit on opposite sides of the male rotor **14** between the teeth **42, 44** while still satisfying the phase-offset relationship with the minimum phase-offset multiplier of two (2).

For balancing the male rotor **14,** it is preferable to have equal radial spacing of the teeth. An even number of teeth is not necessary because an odd number of teeth could also be equally spaced around male rotor **14.** Additionally, the number of teeth that can fit around male rotor **14** is not particularly limited by the preferred embodiment. Generally, arc angle β is proportionally greater than arc angle α according to the phase-offset multiplier. Accordingly, arc angle β of the toothless sector **46** can decrease proportionally to any decrease in the arc angle α of the teeth **42, 44,** thereby allowing more teeth to be added to male rotor **14** while maintaining the phase-offset relationship. Whatever the number of teeth on the male rotor **14,** the female rotor has a corresponding number of helical grooves. Accordingly, the helical grooves **38, 40** have a phase-offset aspect corresponding to that of the helical threads **34, 36.** Therefore, the female rotor has the same number of helical grooves **38, 40** as the number of helical threads **34, 36** on the male rotor, and the helix angle of the helical grooves **38, 40** is opposite-handed from the helix angle of the helical threads **34, 36.**

In the preferred embodiment, each of the helical grooves **38, 40** preferably has a cut-back concave profile **48** and corresponding radially narrowing axial, widths from locations between the minor diameter **50** and the major diameter **52** towards the major diameter **52** at the periphery of the female rotor **16.** The cut-back concave profile **48** includes line segment **jk** radially extending between the minor diameter **50** and the major diameter **52** on a ray from axis **28**, line segment **lm** radially extending between the minor diameter **50** and the major diameter **52,** and a minor diameter arc **lj** circumferentially extending between the line segments **jk, lm.** Line segment **jk** is substantially perpendicular to major diameter **52** at the periphery of the female rotor **16,** and line segment **lmn** preferably has a radius **lm** combined with a straight segment **mn.** In particular, radius **lm** is between straight segment **mn** and minor diameter arc **lj** and straight segment **mn** intersects major diameter **52** at an acute exterior angle ϕ, resulting in a cut-back angle Φ defined by equation (2) below.

$$\text{Cut-Back Angle } \Phi \quad = \quad \text{Right Angle } (90°) - \text{Exterior Angle } \phi, \quad (2)$$

The cut-back angle Φ and the substantially perpendicular angle at opposite sides of the cut-back concave profile **48** result in the radial narrowing axial width at the periphery of the female rotor **16.** In the preferred embodiment, the helical grooves **38, 40** are opposite from each other about axis **28** such that line segment **jk** for each of the pair of helical grooves **38, 40** is directly in-line with each other through axis **28**. Accordingly, in the preferred embodiment, line segment **kjxj'k'** is straight.

In the preferred embodiment of the present invention, the screw rotor device **10** operates as a screw compressor on a gaseous working fluid. Each of the helical threads **34, 36** may also include a distal labyrinth seal **54**, and a sealant strip **56** may also be wedged within the distal labyrinth seal **54**. The distal labyrinth seal **54** may also be formed by a number of striations at the tip of the helical threads (not shown). When operating as a screw compressor, the screw rotor device **10** preferably includes a valve **58** operatively communicating with the outlet port **20**. In the preferred embodiment, the valve **58** is a pressure timing plate **60** attached to and rotating with the male rotor **14** and is located between the male rotor **14** and the outlet port **20**. As particularly illustrated in Figure 4, the pressure timing plate **60** has a pair of cutouts **62, 64** that sequentially open to the outlet port **20**. Between the cutouts **62, 64,** the pressure timing plate **60** forms additional boundaries **66, 68** to the spaces **39, 41** respectively. As the male rotor **14** counter-rotates with the female rotor **16,** boundaries **66, 68** cause the volume in the spaces **39, 41** to decrease and the pressure of the working fluid increases. Then, as the cutouts **62, 64** respectively pass over the outlet port **20**, the pressurized working fluid is forced out of the spaces **39, 41** and the spaces **39, 41** continue to decrease in volume until the bottom of the respective helical threads **34, 36** pass over the outlet port.

Figure 5 illustrates an alternative embodiment of the screw rotor device **10** that only has one helical thread **34** intermeshing with the corresponding helical groove **38** and preferably has a valve **58** at the outlet port **20**. As illustrated in Figure 5, the valve **58** can be a reed valve **70** attached to the housing **12**. In this embodiment, weights may be added to the male

rotor **14** and the female rotor **16** for balancing. The helical groove **38** can have the cut-back concave profile **48** described above, and the male rotor **14** again counter-rotates with respect to the female rotor **16**.

The alternative embodiment also illustrates another aspect of the screw rotor device **10** invention. In this embodiment, the length of the screw rotor device **10** is limited to a single pitch of the helical thread **34** and groove **38**. The pitch of a screw is generally defined as the distance from any point on a screw thread to a corresponding point on the next thread, measured parallel to the axis and on the same side of the axis. The particular screw rotor device **10** illustrated in Figure 5 has a single thread **34** and corresponding groove **38**. Therefore, a single pitch of the **34** and groove **38** requires a complete 360° helical twist of the thread **34** and corresponding groove **38**. . The present invention is directed toward screw rotor devices **10** having the identical number of threads and grooves (N), and the helical twist required to provide the single pitch is merely defined by the number of threads and grooves (N = 1, 2, 3, 4, ...) according to equation (3) below.

$$\text{Single Pitch Helical Twist} \quad = \quad 360°/N \qquad (3)$$

Of course, it will be appreciated that although the length of the screw rotor device **10** is limited to a single pitch, the pitch length can be changed by altering the helix angle of the threads and grooves. The pitch length increases as the helix angle steepens. The screw rotor device **10** illustrated in Figure 1 has a pair of threads **34, 36** and a corresponding pair of helical grooves **38, 40** (N=2). Therefore, a single pitch of these rotors would only require a 180° helical twist (360°/2). However, it is evident that the screw rotor device **10**, as illustrated in Figure 1, has a length slightly greater than two pitches. Therefore, for the given length of the rotors, the helix angle for the threads and grooves would have to increase for the rotors to have a single pitch length. For example, Figures 7A and 7B illustrate a screw rotor device **10** that has a pair of threads **34, 36** and a corresponding pair of helical grooves **38, 40** that are limited to a 180° helical twist. Accordingly, Figures 7A and 7B particularly illustrate rotor lengths that are limited to the single pitch of the threads **34, 36** and grooves **38, 40**.

The screw rotor device **10** illustrated in Figure 7A also incorporates the phase-offset relationship into its design. The angle between ray segment **oa** and ray segment **ob,** subtending tooth **42,** is arc angle α. According to the phase-offset definition provided above, arc angle β of the toothless sector **46** must extend from ray segment **ob** to at least to ray segment **oa',** which would correspond to twice the arc of arc angle α, the minimum phase-offset multiplier being two (2) in equation 1.

As particularly illustrated in Figure 3, the helical thread **34** preferably has a cut-in convex profile **72** that meshes with the cut-back concave profile **48** of the helical groove **38**. The cut-in convex profile **72** has a tooth segment **74** radially extending from minor diameter arc **ab**. The tooth segment **74** is subtended by arc angle α and is further defined by equation (4) below according to arc angle θ for minor diameter arc **ab.**

$$\text{Arc Angle } \alpha \qquad > \qquad \text{Arc Angle } \theta \qquad\qquad (4)$$

The phase-offset relationship defined for a pair of threads is also applicable to the male rotor **14** with the single thread **34,** such that the toothless sector **46** must have an arc angle β that is at least twice the arc angle α of the single helical thread **34.** The male rotor **14** circumference is 360°. Therefore, arc angle β for the toothless sector **46** must at least 240° and arc angle α can be no greater than 120°. Similarly, for the pair of threads **34, 36,** 60° is the maximum arc angle α that could satisfy the minimum phase-offset multiplier of two (2) and 30° is the maximum arc angle α that could satisfy the phase-offset multiplier of five (5) for the preferred embodiment. For practical purposes, it is likely that only large diameter rotors would have a phase-offset multiplier of 50 (3° maximum arc angle α) and manufacturing issues may limit higher multipliers.

The male rotor **14** and female rotor **16** each has a respective central shaft **76, 78.** The shafts **76, 78** are rotatably mounted within the housing **12** through bearings **80** and seals **82.** The male rotor **14** and female rotor **16** are linked to each other through a pair of counter-rotating gears **84, 86** that are respectively attached to the shafts **76, 78**. The central shaft **76** of the male rotor **14** has one end extending out of the housing **12.** When the screw rotor device **10** operates as a

compressor, shaft **76** is rotated causing male rotor **14** to rotate. The male rotor **14** causes the female rotor **16** to counter-rotate through the gears **84, 86,** and the helical threads **34, 36** intermesh with the helical grooves **38, 40.**

As described above, the distal labyrinth seal **54** helps sealing between each of the helical threads **34, 36** on the male rotor **14** and the cylindrical bore **30** in the housing **12.** Similarly, as particularly illustrated in Figure 3, axial seals **88** may be formed in the housing **12** along the length of the cylindrical bore **32** to help sealing at the periphery of the female rotor **16.** As the male rotor **14** and female rotor **16** transition between meshing with each other and respectively sealing around the housing **12**, a small gap **90** is formed between the male rotor **14**, the female rotor **16** and the housing **12**. The rotors **14, 16** fit in the housing **12** with close tolerances.

As discussed above, the preferred embodiment of the screw rotor device **10** is designed to operate as a compressor. The screw rotor device **10** can be also be used as an expander. When acting as an expander, gas having a pressure higher than ambient pressure enters the screw rotor device **10** through the outlet port **20**, valve **58** being optional. The pressure of the gas forces rotation of the male rotor **14** and the female rotor **16**. As the gas expands into the spaces **39, 41,** work is extracted through the end of shaft **76** that extends out of the housing **12**. The pressure in the spaces **39, 41** decreases as the gas moves towards the inlet port **18** and exits into ambient pressure at the inlet port **18**. The screw rotor device **10** can operate with a gaseous working fluid and may also be used as a pump for a liquid working fluid. For pumping liquids, a valve may also be used to prevent the fluid from backing into the rotor.

Figures 6A and 6B illustrate a detailed cross-sectional view of the helical grooves and helical threads from Figures 2A and 2B, respectively. These views illustrate the differences between an acme thread profile **92** and another feature of the present invention, a buttress thread profile **94**. Between the minor diameter **50** and the major diameter **52** of the female rotor, the acme thread profile **92** of the helical groove **38** includes a concave line **96** and a substantially straight line **98** opposite therefrom. The buttress thread profile **94** also includes a concave line **96**

but is particularly defined by a diagonal straight line **100**. On the male rotor, the acme thread **92** profile of the helical thread **34** is also between the major and minor diameters and includes a pair of opposing convex curves. In comparison, the buttress thread profile **94** has a diagonal straight line **102** that is parallel to and in close tolerance with the corresponding diagonal straight line **100** in the helical groove **38**. In the particular example illustrated by Figure 6B, a convex curve **104** is opposite the diagonal straight line **102**.

Figures 7A and 7B particularly illustrate the screw rotor device **10** according to several aspects of the present invention, including the parallel diagonal straight lines **100, 102** of the buttress thread profile **94**, phase-offset helical threads **34, 36**, and the single pitch design of the male and female rotors **14, 16** within the housing **12**. With regard to the particular example illustrated by Figure 7B, the buttress thread profile **94** includes a concave curve **104** opposite from the diagonal straight line **102**. It should be appreciated that the benefits of the present invention can be achieved with manufacturing tolerances, such as in the parallel diagonal straight lines **100, 102**. In particular, tolerances in the parallel diagonal straight lines **100, 102** may allow for a slight radius of curvature between the diagonal lines and the major and minor diameters and an extremely slight divergence in the parallelism. It will be appreciated that manufacturing tolerances may vary depending on the type of material being used, such as metals, ceramics, plastics, and composites thereof, and depending on the manufacturing process, such as machining, extruding, casting, and combinations thereof.

In view of the foregoing, it will be seen that the several advantages of the invention are achieved and attained. The embodiments were chosen and described in order to best explain the principles of the invention and its practical application to thereby enable others skilled in the art to best utilize the invention in various embodiments and with various modifications as are suited to the particular use contemplated. As various modifications could be made in the constructions and methods herein described and illustrated without departing from the scope of the invention, it is intended that all matter contained in the foregoing description or shown in the accompanying

drawings shall be interpreted as illustrative rather than limiting. Thus, the breadth and scope of the present invention should not be limited by any of the above-described exemplary embodiments, but should be defined only in accordance with the following claims appended hereto and their equivalents.